Thermal Analysis of Ballscrew Systems by Finite Difference Methods

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INTRODUCTION

Ballscrews have high stiffness, long life, good efficiency, and low driving torque. It is used as a general positioning element in machine tools and precision machinery. To compensate for the positioning error, and to predict accuracy of precision machines during the design process, accurate deformation estimates of the feed drive system equipped with ballscrews are required. In addition to contact and body deformation occurred in the static state [1], thermal deformation due to friction between balls and grooves degrades positioning accuracy of ballscrew driven feed drive systems. To calibrate the thermal deformation error, temperature measurement of the ballscrew is required. However, it is extremely inefficient and almost impossible to acquire the whole temperature distribution by measuring temperatures of every point. Therefore, a thermal model estimating the whole temperature field from mathematical model is required [2-5].

In this paper, contrary to previous researches [4-5], heat generation mechanism of ballscrews and temperature analysis module are more accurately derived through stable numerical analysis of heat transfer. To improve stability and computational efficiency of the finite difference method, temperature distribution is estimated through the alternating direction implicit (ADI) method. Validity of the developed thermal analysis method is to be verified through experiments on the ballscrew driven feed drive system under various operating conditions.

THERMAL ANALYSIS

Figure 1 shows a double-nut ballscrew feed drive system. As the ballscrew rotate, heat is generated because of slip between balls and grooves, viscous friction due to shear stress of lubricant, and rolling contact friction. Support bearings generates frictional heat as well. In addition, environmental temperature variation as an external heat source changes temperature distribution of ballscrews.

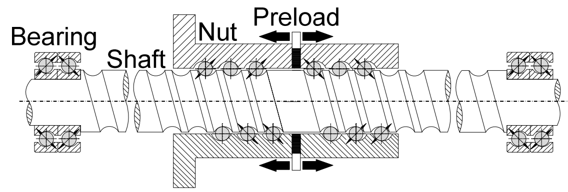

FIGURE 1. Double-nut ballscrew assembly.

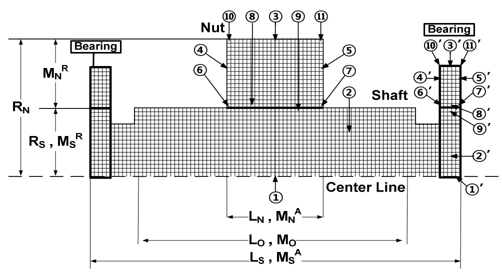

FIGURE 2. Axisymmetric finite difference model.

Alternating Direction Impicit (ADI) Method

Fig. 2 shows a finite difference model with smaller axial and radial difference △z and △r, respectively. Implicit finite difference heat equation for the nodes at region ② is given by

$$T_{m,n}^{\;i+1/2} = \frac{\alpha\Delta t}{2(\Delta r)^2}\left[\left(\frac{r_{m+1/2}}{r_m}\right)T_{m+1,n}^{\;i+1/2} + \left(\frac{r_{m-1/2}}{r_m}\right)T_{m-1,n}^{\;i+1/2} - 2T_{m,n}^{\;i+1/2}\right] + \frac{\alpha\Delta t}{2(\Delta z)^2}\left(T_{m,n+1}^{\;i} + T_{m,n-1}^{\;i} - 2T_{m,n}^{\;i}\right) + T_{m,n}^{\;i} \tag{1}$$

$$T_{m,n}^{\;i+1} = \frac{\alpha\Delta t}{2(\Delta r)^2}\left[\left(\frac{r_{m+1/2}}{r_m}\right)T_{m+1,n}^{\;i+1/2} + \left(\frac{r_{m-1/2}}{r_m}\right)T_{m-1,n}^{\;i+1/2} - 2T_{m,n}^{\;i+1/2}\right] + \frac{\alpha\Delta t}{2(\Delta z)^2}\left(T_{m,n+1}^{\;i+1} + T_{m,n-1}^{\;i+1} - 2T_{m,n}^{\;i+1}\right) + T_{m,n}^{\;i+1/2}$$

where $T_{m,n}^{\;i+1}$ and $T_{m,n}^{\;i}$ are next and present step temperatures, respectively. Considering heat balance condition, implicit form of finite difference heat equations for 22 parts of the FDM model are formulated according to boundary conditions. Heat convection, ambient temperature variation, and frictional heat generation from internal heat sources such as support bearings and contact surfaces between balls and grooves are to be included in the models.

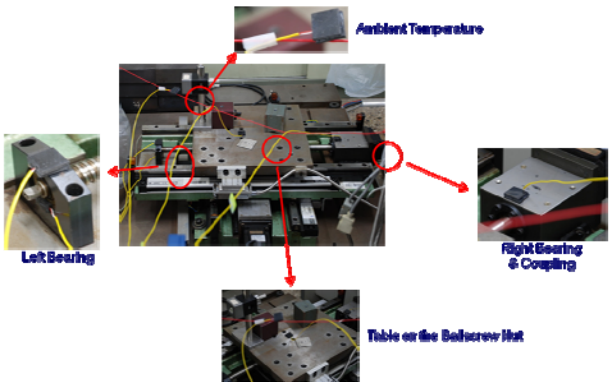

FIGURE 3. Experimental setup.

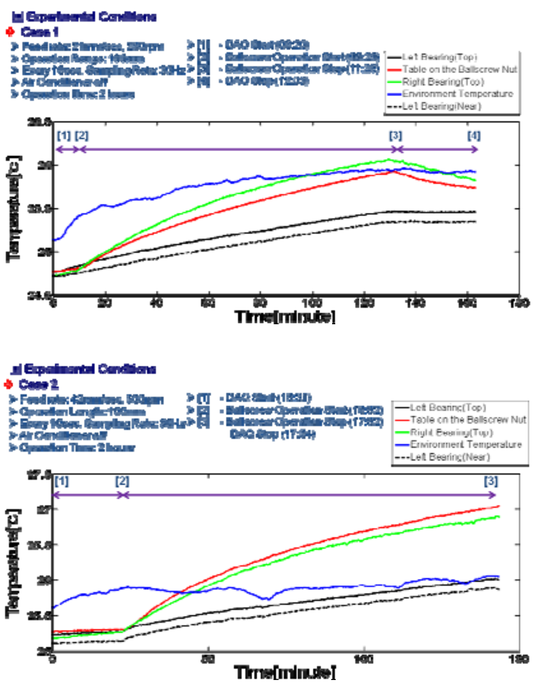

FIGURE 4. Temperature measurements.

Convection Coefficient

To estimate temperature distribution accurately, convection heat transfer coefficients along the ballscrew and nut surfaces should be selected properly. It depends upon air flow speed on objects as given by Eq. (2).

$$h = c\,\mathrm{Re}^{m}\,\mathrm{Pr}^{1/3} \qquad (2)$$

where Re is the Reynolds number. c and m are subject to Re, and Pr is the Prandtl number.

Heat Generation Mechanism

Frictional torques generated from balls and grooves are calculated by applied load, viscosity and spinning motion. Previous estimation methods [5] with better convection coefficients are to be applied for the thermal analysis.

EXPERIMENTAL VERIFICATION

Fig. 3 shows experimental setup to verify developed thermal analysis FDM module. Fig. 4 shows temperature measurement results at different operating conditions. Through repetitive measurements of frictional torques and ambient temperatures, and applying them to numerical simulations of the FDM model, we can find good boundary conditions for the FDM analysis. Proper thermal analysis module for a ballscrew driven feed drive system can be proposed in this study.

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